Development of a High Frequency Pulse Tube

N. Emery[1], A. Caughley[1], N. Glasson[1], A. Tucker[2], M. Gschwendtner[3]

[1]Industrial Research Ltd., Christchurch, New Zealand
[2]University of Canterbury, Christchurch, New Zealand
[3]TS-dot Engineering Ltd., Christchurch, New Zealand

ABSTRACT

A single-stage pulse tube cryocooler has been designed and fabricated to provide cooling at 50 K for a high temperature superconducting magnet. Sage software was used to design the pulse tube, with a predicted 25 W of cooling power at 50 K, and an input indicated power of 1.7 kW. The fabricated pulse tube is closely coupled to a metallic diaphragm compressor (pressure wave generator) with a 60 ml swept volume, operating at 50 Hz, and with a mean helium working gas pressure of 25 bar. Details of the development, experimental results and correlations to the Sage model are discussed.

INTRODUCTION

Industrial Research Ltd. (IRL) began research in cryogenic refrigeration in 2004 with a project objective of creating an industrialized cryocooler for cooling High Temperature Superconducting (HTS) applications such as transformers and magnets. IRL's initial work in the cryogenic refrigeration program concentrated on the compressor side of the refrigerator. This led to a novel pressure wave generator (PWG) technology using metallic diaphragms[1] to separate the clean cryocooler working gas from a conventionally lubricated driving mechanism. The first 200 ml swept volume PWGs successfully powered a number of pulse tube refrigerators[1]. IRL made a pulse tube for a 20 ml swept volume PWG that achieved 7W of refrigeration at 77K. This paper describes the design, manufacture and experimental optimization of a pulse tube cryocooler. The planned outcome is to produce a pulse tube cryocooler capable of cooling an HTS magnet made by HTS-110 Ltd. The cooling requirement for the magnet is 20 W @ 50 K hence this pulse tube was designated the PT2050.

NUMERICAL SIMULATION

A numerical 1-D model of a pulse tube was created using the pulse tube version of the Stirling (regenerative cycle) simulation software Sage[2]. The pulse tube was coupled to a PWG with 60 ml of swept volume. A cross-section of the pulse tube is shown in Figure 1. The Sage model for the 7 W at 77 K pulse tube[3] made by IRL was used as a starting point. The initial geometric parameters were based on general relationships[4,5] such as the cross-sectional area of the pulse tube is proportional to the cooling power requirement, the pulse tube volume is approximately 0.4 times the compressor

Cryocoolers 16, edited by S.D. Miller and R.G. Ross, Jr.

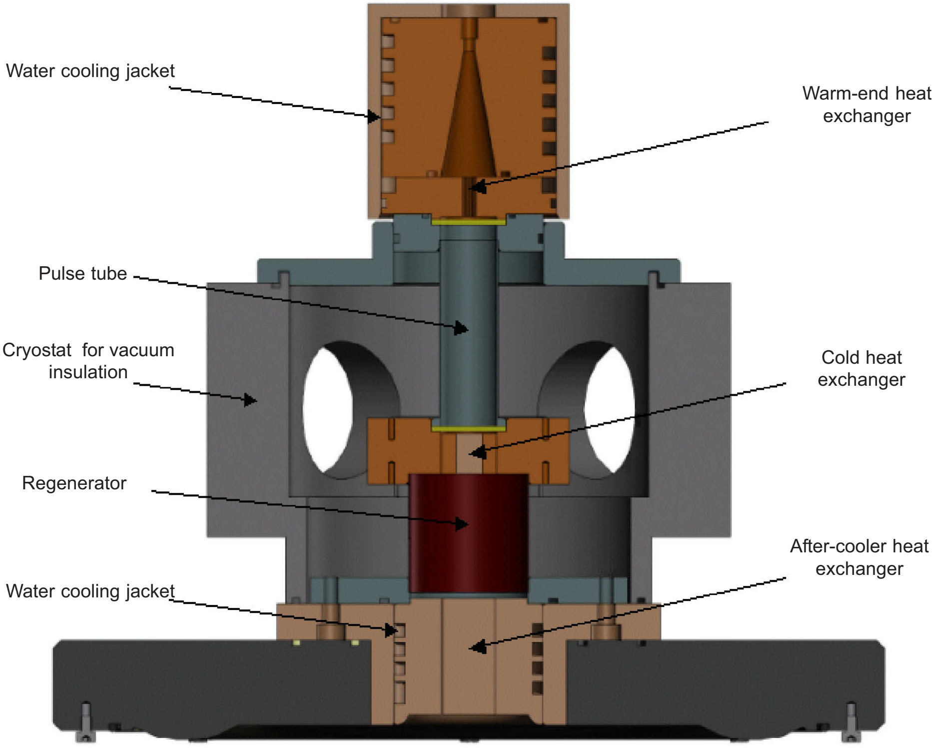

Figure 1. Cross-sectional view of the PT2050 Pulse Tube.

swept volume and the reservoir is approximately 50 times the pulse tube volume. It was found that after optimization, the final model was near the initial conditions mentioned above.

Improvements were made to the initial Sage model, for example, changing the internal heat exchangers from channel type to fin type, cross-referencing various outputs such as heat-exchanger cross-sectional areas in the output files to calculated areas to account for the tapered fins and changing surfaces to suit thermal wavelength versus wall thickness as per the Sage 6 manual[2]. A sensitivity analysis was conducted. It was found that the model was sensitive to N cells (mesh size parameter) in the pulse tube, which was mapped to determine a value close to the asymptotic value, while remaining computationally efficient. Convergence times with the model were reasonable at around 30 seconds per run. Computation time with approximately 5 parameters to map was up to 15 minutes. It was found that the sensitivity of various parameters in the model can change if, for example, another parameter is added, or even changed. An attempt has been made to find the interdependency of each parameter within the model.

Internally slotted water-cooled heat-exchangers were used to transfer heat at the aftercooler, cold head and warm end of the pulse tube. There were three heat flows to consider when designing the heat-exchangers: heat transfer from the working gas to the metal; conduction down the fins to the cooling water channels, and heat transfer to the water. The greatest resistance to heat flow was found to be between the working gas and the internal fins.

The heat-exchanger geometry and associated nomenclature are shown in Figure 2. Sage model mappings showed that for a given cross-sectional flow area (H x W x n (number of slots)), the thinnest slots (W) gave the best cooling performance, if H and n were optimized. It was found that several compromises were necessary. Thinner slots increased the pressure drop across the heat-exchanger (negative), heat transfer increased (positive), dead volume decreased (positive), and the cost of wire-cutting operation increased (negative). The minimum slot width was eventually determined by practical wire cutting limitations. The heat-exchanger length (L) also contributed to the

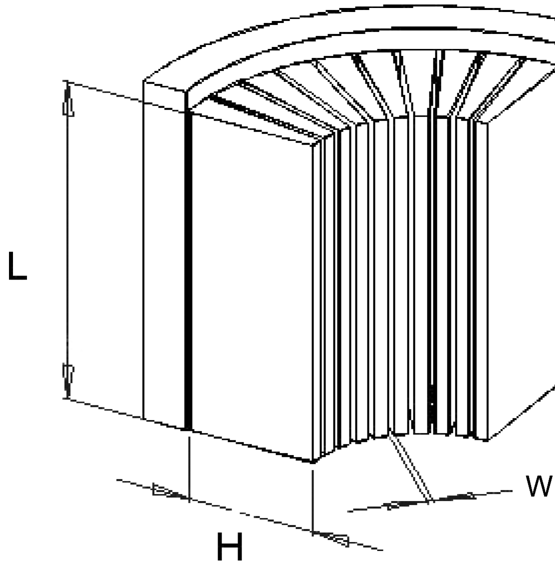

Figure 2. Heat-exchanger fin slot geometry

flow result, with the longer lengths (L) requiring greater flow area. This makes sense as there is a greater pressure drop due to the longer length and increasing the flow area reduces the pressure drop.

Sage was used to optimize the gas transfer geometry and Cosmos finite element analysis (FEA) was used to look at heat conduction down the fins. A variety of fin configurations were analyzed for efficient conduction such as a tree type arrangement, tapered fins and different fin angles. The taper on the fins allowed a more uniform heat flux down the fins as they gained (or gave) energy from the flow. The highest heat conduction was found for the thinnest slot (W) due to the larger quantity of metal that remained for conduction. It was found that radial slots gave the best result. A relatively large centre hole was required to accommodate the number of slots and slot width yet allow enough fin material to hold a plug. Figure 3 shows the copper warm end heat-exchanger that was used in the pulse tube.

Figure 3. Warm heat-exchanger

Assumptions made within the Sage model include: the water-cooling to the warm heat-exchanger and aftercooler provided an isothermal surface: 3-D flow losses throughout the pulse tube were assumed negligible; Radial heat transfer in the regenerator mesh is negligible; Thermal streaming in the regenerator is negligible; Acoustic streaming in the pulse tube is negligible. Some of these assumptions may not be correct, and together are likely to predict a better performance than observed. Moreover, the output cooling power in the Sage simulation did not include heat leaks down sensor wires, heat leaks from conduction through gas (non ideal vacuum), heat leaks from conduction along the MLI, radiation heat leaks and radial conduction through the regenerator. These have been experimentally estimated and are discussed in the results section of this paper.

The final SAGE model predicted 35 W of cooling power at 50 K for a PV power input of 1500 W.

DESIGN AND BUILD

The pulse tube was manufactured in Christchurch, New Zealand using standard industrial equipment. Figure 4 shows a 3-D model of the PT2050 coupled to the CHC60 was created using Solidworks. The model was created using the geometries obtained from the Sage model. The pulse tube sits directly on top of the pressure wave generator's top plate with the entrance of the copper after-cooler as close as possible to the diaphragm. The CHC60 pressure wave generator incorporates the new hydraulic amplifier drive mechanism and a cast crankcase.

Heat Exchangers

The heat exchangers were machined from copper with wire cut internal fins. A central plug was shrunk with liquid nitrogen before fitting into the center of the heat exchangers. The fins slots were 0.2 mm wide.

Regenerator

The regenerator was created from 794 stainless steel, 400 mesh discs laser cut from sheet. The number of discs required was calculated from the diameter of two of the wires in the mesh (0.03 mm) and the length of space available for the mesh. The mesh discs were weighed with high accuracy

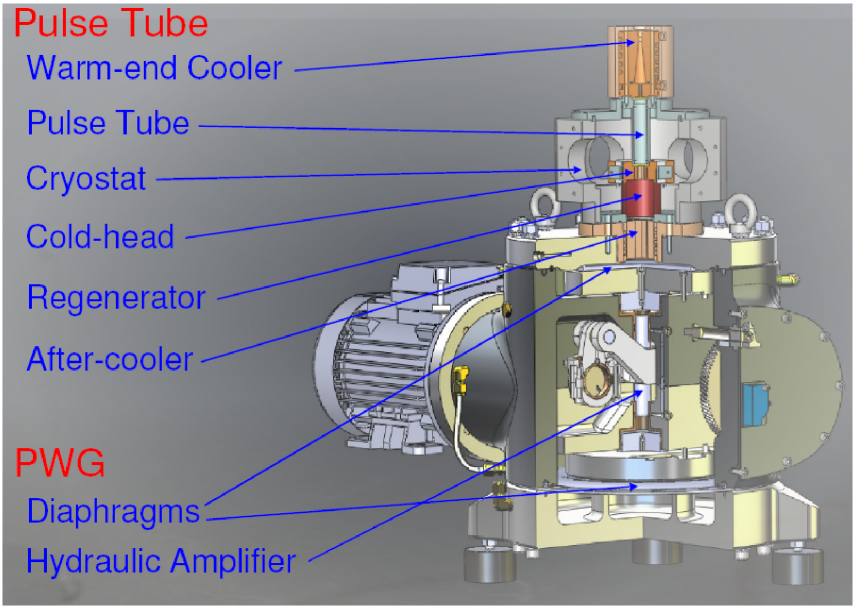

Figure 4. Cryocooler - CHC60 PWG and PT2050 Pulse Tube

scales to determine the number of discs to assemble. The discs were then assembled in random orientations (not at 45 degrees as would be potentially better). A press was used to compress the discs to make the assembly easier. Stainless steel tubing was used to house the mesh. The tube was 47 mm long with a 44 mm inside diameter, and had a 0.4 mm wall thickness to reduce thermal losses down the length of the tube.

Pulse Tube

The pulse tube was fabricated from a standard stainless steel tube with internal diameter of the correct size, so only the outside diameter required machining. The pulse tube was 76 mm long with a wall thickness of 0.4 mm and inside diameter of 21 mm.

Inertance Tube

A copper inertance tube with an inside diameter of 4.3 mm and a 1 mm wall thickness was used with lengths ranging between 2 m and 4 m.

Reservoir

An existing 2.3 liter reservoir was used for experiments. The eventual aim is to use the PWG's gas spring as the reservoir.

Flow Straighteners

Brass 100 mesh discs were used with a small plenum at each end of the pulse tube to help straighten the flow of gas in the pulse tube.

Assembly

The components were either soldered or bolted together with O-rings to seal in the helium working gas. The pulse tube was polished and washed in acetone in an ultrasonic cleaner. The mesh discs were also washed in acetone. The acetone was baked out, and a vacuum applied.

Instrumentation

Cold-head temperature was measured with a Lakeshore silicon diode[6] and a PT100 platinum sensor. Both were four-wire measurements thermally anchored by winding the leads up the pulse tube. The details are seen in Figure 5. A Lakeshore temperature monitor was used to measure the

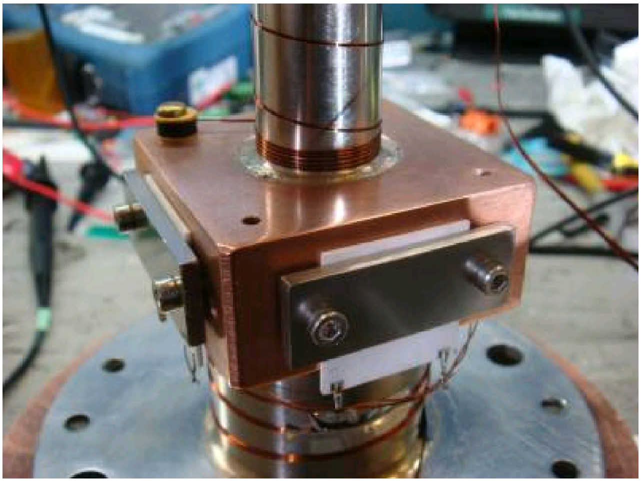

Figure 5. PT2050 Pulse Tube wired with heaters and temperature sensor

cold temperatures. Cooling power was measured by applying power to four 100 Ω resistors clamped to the cold-head. A quick secondary cooling power measurement was made by recording the cool down rate of the cold heat exchanger's copper block, whose thermal mass dominated the pulse tube. The two methods agreed with each other. The sensors provided inputs to a National Instruments SCXI signal conditioner and were processed on computer. The data acquisition program provided a real time display as well as recording a data log for post-run analysis.

Insulation

Surfaces were polished, then heat shielded with MLI to reduce radiation losses. A vacuum of 10^{-6} mbar was achieved.

RESULTS AND DISCUSSION

The pulse tube was initially run on an oversized compressor (until the 60 cc compressor was available), which enabled some initial tuning and instrumentation to be conducted. A buffer volume was inserted between the CHC240 (240 cc) PWG[3] and the pulse tube to reduce the pressure ratio. Even so, a low charge pressure was necessary to limit the magnitude of the pressure wave and avoid damage to the PWG drive. The lowest no-load temperature of the pulse tube on the CHC240 PWG was 49.6 K. The optimum mean gas pressure was 9 bar which gave a pressure ratio of 1.9. The optimum frequency was 42 Hz with a 4 m inertance tube. 100 W of cooling power was produced at 105 K with 20 bar mean gas pressure.

A 9.5 mm diameter orifice valve (ball valve) was used as the variable phase shifter, as a starting point in characterizing the performance of the pulse tube on the CHC60 (60 cc) PWG. The orifice valve was placed after the transition cone at the warm end of the pulse tube. A 2m long 9.5 mm diameter inertance tube was placed between the orifice valve and the reservoir. Frequencies of 50 and 60 Hz were run with no-load temperatures and 25 W temperatures, as shown in Table 1. The valve opening was adjusted for each change in parameter. The best no-load frequency was higher than 60 Hz with this phase shifter and average gas pressure.

A set of 4.3 mm diameter inertance tubes of different lengths were tested. One of the big advantages of the PWG used for the experiments was that the frequency of operation could be adjusted over a wide range to gain a feel for which direction the inertance tube length needed to go, and by roughly how much. So, for example, if the required frequency is 50 Hz and a 2 m inertance tube gave its best performance at 60 Hz, then one could predict going longer in the tube to bring the tuned frequency down. If next a 4 m tube was found to be best at 40 Hz, then the obvious next tube size might be a 3 m length.

Heat leak was measured using a procedure[5] that involved measuring the warm up rate at a set temperature with a range of heat loads applied to the cold-head. Extrapolation back to a zero warm up rate gives the heat leak into the cold head. The heat-up rate was recorded for each of the three powers at 120K, and then plotted as is shown in Figure 6. A thermal loss of just under 10 W is shown, which includes such things as working gas conduction, regenerator mesh conduction, pulse

Table 1. Orifice valve performance

Frequency	Avg. Pressure	Pressure ratio	Input PV	PV angle	Load	Warm Temp.	Cold Temp.
Hz	Bar		W	Degrees	W	K	K
40	26.12	1.377	767	149.6	25	290	103.2
50	26.21	1.387	1001	149.2	0	290	88.5
50	26.23	1.395	948	151.4	25	290	102.6
60	26.19	1.413	1254	148.0	0	290	87.6
60	26.20	1.414	1181	149.9	25	290	99.5

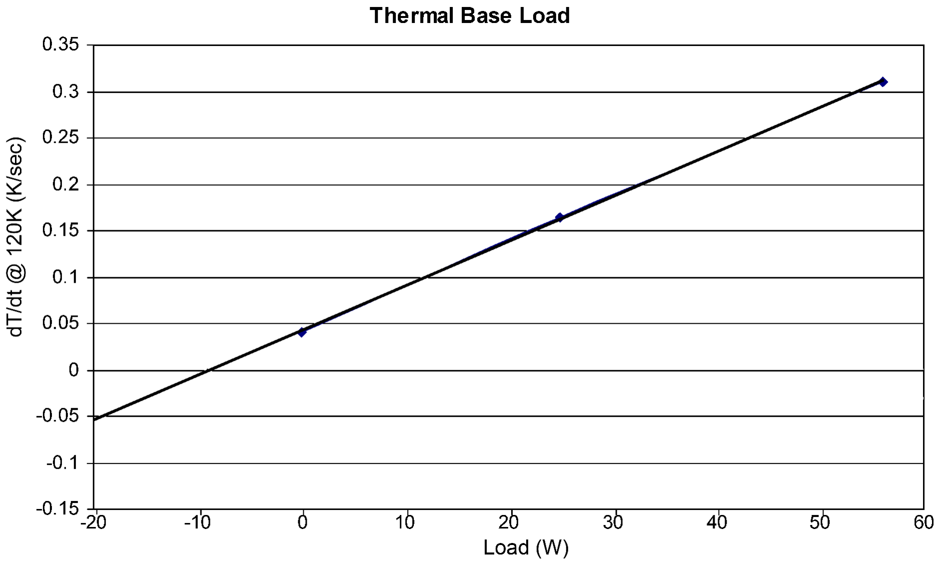

Figure 6. Warm up rate for different thermal heat inputs at 120K.

tube conduction, regenerator tube conduction, sensor wire conduction, heater wire conduction, cryostat radiation, cryostat gas conduction and MLI conduction.

The Sage model had improvements made, as well as was updated to reflect changes to components and features during manufacture and assembly. The updated model now predicts cooling power to be 35 W @ 50 K as compared to the original prediction of 25 W @ 50 K. However the pressure ratios at various temperatures correlated better. The slope trends of the cooling power curves at various pressures also correlate better. However the cooling power with the CHC60 has been measured to be much less than the predictions, with no-load temperatures between 60 and 65 K. The reduced cooling power observed is reflected by a lower than predicted PV power. Work is in progress to compare several of the experimental outputs to the Sage model, now that we have a working pulse tube.

Experiments were conducted to determine the relationship between cooling performance and mean gas pressure. The experimental results have been compared to the Sage model. Figure 7 shows the effect of mean gas pressure on cooling power at a temperature for both the Sage model and the experiment. A 4.3 mm diameter, 4 m inertance tube length was used, and the frequency of operation was optimized in the experiment to give the best no-load temperature. It was found that lower mean gas pressure gave flatter cooling power vs. temperature curves, but in most cases either the same or better no-load temperatures.

CONCLUSIONS

A pulse tube was designed and fabricated. It has achieved a lowest no-load temperature of 49.6 K and produced 100 W of cooling power at 105 K on a CHC240 PWG. The smaller CHC60 PWG resulted in a lower PV power and had a correspondingly higher no-load temperature of 59.7 K and 100 W of cooling power at 128 K. Testing indicates that the mean gas pressure has a significant effect on the cooling power. It has been observed that lower pressures can provide lower no-load temperatures from the configurations trialed to date.

Some correlation has been found between the Sage model and the experiment, such as the effect of mean gas pressure and the slope of a particular cooling curve. Work is continuing to understand the differences between the Sage model and the experiment in order to more efficiently design PWG – pulse tube systems.

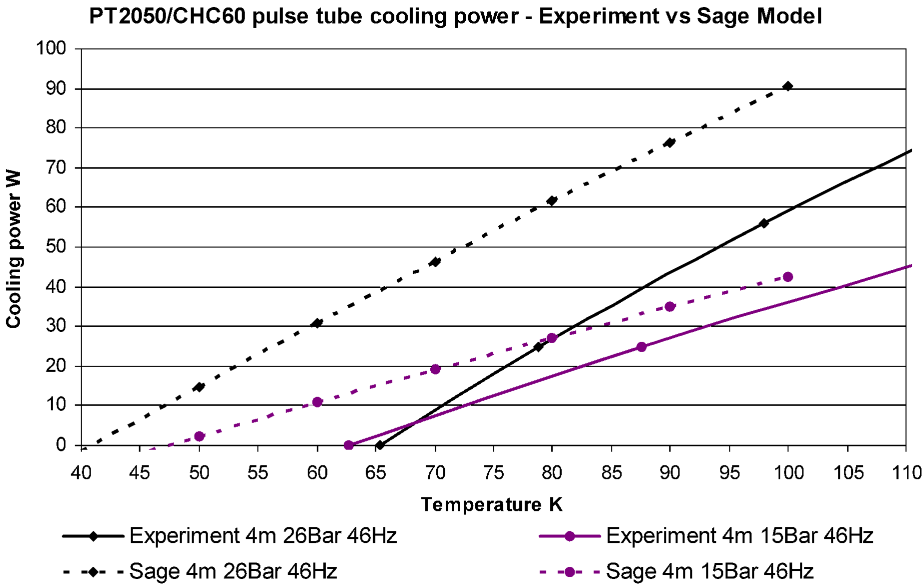

Figure 7. Cooling power graph comparing mean gas pressure - at 46 Hz with a 4 m inertance tube

ACKNOWLEDGMENT

The authors acknowledge Industrial Research Ltd., New Zealand for continued support of this work, University of Canterbury, New Zealand's Foundation for Research, Science and Technology for funding, David Gedeon for his Sage software and excellent support, Mace Engineering for their manufacturing assistance and HTS-110 for supporting the commercialization of the device.

REFERENCES

1. Caughley, A.J. et al, "A Low Cost Pressure Wave Generator using Diaphragms," *Adv. in Cryogenic Engineering*, Vol. 53, Amer. Institute of Physics, Melville, NY (2008), pp.1122-1129.
2. Gedeon, D., *Sage User's Guide*, Gedeon Associates (2009).
3. Caughley, A.J. et al, "Diaphragm Pressure Wave Generator Developments at Industrial Research Ltd," *Adv. in Cryogenic Engineering*, Vol. 55, Amer. Institute of Physics, Melville, NY (2010), pp. 695-702.
4. Radebaugh, R., Foundation of Cryocoolers Short Course, Cryogenic Society of America (2004).
5. Yuan, J., Maguire, J., "Development of a Single Stage Pulse Tube Refrigerator with Linear Compressor," *Cryocoolers 13*, Kluwer Academic/Plenum Publishers, New York (2005), pp.157-163
6. Lakeshore Cryotronics, Silicon Diode Installation Instructions, 13 January 2010.